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Research article

Application and analysis of a model based controller for cooling towers in compression chiller plants

Parantapa Sawant [a,*], Eric Ho [b], Jens Pfafferott [a]

[a] Institute of Energy Systems Technology (INES), Offenburg University of Applied Sciences, Offenburg, Germany
[b] Department of Mechanical and Aerospace Engineering, Carleton University, Ottawa, Canada

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ABSTRACT

Cooling towers or recoolers are one of the major consumers of electricity in a HVAC plant. The implementation and analysis of advanced control methods in a practical application and its comparison with conventional controllers is necessary to establish a framework for their feasibility especially in the field of decentralised energy systems. A standard industrial controller, a PID and a model based controller were developed and tested in an experimental set-up using market-ready components. The characteristics of these controllers such as settling time, control difference, and frequency of control actions are compared based on the monitoring data. Modern controllers demonstrated clear advantages in terms of energy savings and higher accuracy and a model based controller was easier to set-up than a PID.

1. Introduction

Heating, Ventilation and Air Conditioning (HVAC) systems can contribute up to 42% of the total energy consumption in commercial buildings (Henze et al., 2004). However, studies have demonstrated that appropriate selection and operation of these systems can provide energy savings by up to 25% (Afroz et al., 2018).

One example for potential savings is the electric fan cooling tower or recooler (dry and wet) for the condensers of a chilling machine. They are responsible for high electricity consumption if not designed and controlled efficiently (Sousa et al., 1997).

Classical open-loop On-Off controllers or temperature based closed-loop On-Off controllers are most widely implemented due to their simplicity but they are unable to control moving processes with time delays and can lead to energy wastage especially in transition seasons. Proportional Integral Derivative (PID) controllers have produced promising results but their performance can degrade if the operating conditions of the systems vary from the tuning conditions. Additionally, the tuning of PID controllers can be a cumbersome task and often requires extensive engineering knowledge of the system (Afram and Janabi-Sharifi, 2014). Hard controllers like Model Predictive Control (MPC) demonstrated better performance, robustness and easier implementation in thermally inertial HVAC systems (Khadraoui et al., 2015; Serale et al., 2018).

Table 1 shows selective research work done for controlling only fan-coils and recoolers in the field of HVAC. It is clear that there is a lack of practical implementation, evaluation, and comparison of conventional and modern controllers for HVAC cooling towers. In this paper an experimental comparison of a standard industrial controller, a PID controller, and an open-loop Model Based Controller (MBC) is done. The PID controller is implemented using the PID palette from the "Control and Simulation" toolbox in LabVIEW® and is designed using the Ziegler-Nicholson method. The MBC is implemented using the "Mathscript" module in an iterative loop in LabVIEW® and is based on a static model of the cooling tower. These controllers are applied completely in their theoretical form and no further tuning is done for a direct comparison of the methodologies. This gives a fair idea about the complexity for their set-up in case tuning would be necessary. This work contributes to the state-of-art in research on HVAC control by applying a novel and practical MBC for electrical recoolers and experimentally evaluating its benefits for energy engineers and industrial developers especially in a retrofit scenario using conventional equipment and data acquisition methods.

The sections of this paper are:

* Corresponding author.
E-mail addresses: parantapa.sawant@hs-offenburg.de, jens.pfafferott@hs-offenburg.de (P. Sawant).

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Table 1. Literature on control of fan-coils and cooling towers in building-HVAC systems.

Reference	Controlled System	Controlled Variable	Controller	Research Objective
(Bengea et al., 2012)	HVAC system of a medium-size office building in a field demonstration	Room temperature	MPC to optimise a variable volume, dual-duct, multi-zone HVAC unit	•Demonstration of optimal control •Minimise energy consumption of entire system
(Ma et al., 2009)	Cooling system for a university campus with wet cooling towers, chillers and cold water storage in a simulation environment	Storage temperature	MPC to decide optimal set-point temperatures and water mass flow rates for chiller and cooling tower	•Minimise electricity costs of entire system •Maximise COP of entire system
(Soyguder and Alli, 2009; Hosoz et al., 2011)	HVAC systems for two 0.5 m^3 chambers in a lab set-up	Room temperature	PID and ANFIS to control damper-rates and fan-speed	•Comparison of PID and ANFIS algorithms
(Teitel et al., 2008)	Ventilation fans with a variable-speed drive unit and an On-Off unit for a greenhouse and poultry house in a field demonstration	Room temperature and humidity	On-Off and VFD to control fan-speed	•Comparison of On-Off and VFD algorithms
(Tianyi et al., 2011)	Fan coil units with three speeds and an electric on-off valve for a 10m^2 area in an experimental set-up	Room temperature	DRFCM to control fan-speed and water mass-flow rate	•Minimise energy consumption of entire system
(Wemhoff, 2012)	Two-room HVAC system with a 1.5 kW_{el} chiller and a variable-speed fan in a simulation environment	Room temperature	PID to control different equipment of the HVAC system	•Minimise energy consumption •Study effect of calibration of PID coefficients on energy savings
(Yu and Chan, 2007)	Cooling system with air-cooled chiller and cooling tower with three speeds in a simulation environment	Condenser inlet temperature	MBC to control fan speed of cooling tower	•Maximise COP of the chiller

ANFIS: Adaptive Network Based Fuzzy Inference System, DRFCM: Duty Ratio Fuzzy Control Method, VFD: Variable Frequency Drive.

- Description of controlled system,
- Mathematical model of cooling tower,
- Formulation of the MBC and PID controllers,
- Quantitative and qualitative discussion on the experimental results.

2. Methodology

2.1. Controlled system

At the Institute of Energy Systems Technology (INES) in Offenburg University of Applied Sciences, a microscale trigeneration plant has been installed using standard industrial components (Sawant and Pfafferott, 2017). The components in the scope of this study are the Compression-Chilling Machine (CCM) and its fan based dry Cooling Tower (CT) seen in Figure 1a, b.

The Data Acquisition and Control System (DAQ) is based on an OPC Client/Server network. The OPC server is a Beckhoff CX9020® Programmable Logic Controller (PLC) communicating with the physical components and instruments using MBUS standards at a Baudrate of 2400 ms. The OPC client is the Shared Variable Engine in LabVIEW®. Monitoring data was logged using change-of-value with a logging deadband of 2 % of previous value. The logged data is extracted from the database at a 3 seconds time interval using natural interpolation. The logging resolution of temperature, volume flow and power is 0.1 °C, 0.0 m^3/h and 0.1 kW.

The controlled system is seen in Figure 2, where the CT is the heat sink for the condenser of the chilling machine. It comprises of three variable-speed fan motors consuming a maximum electrical power P_{el_max} of 0.9 kW_{el} at their maximum speed RPM_{max} of 480 RPM. At RPM_{max} the maximum mass flow of air $\dot{m}_{air,max}$ is 46,300 kg/h. The actual speed of the fans RPM can be controlled with a 0–10 volt signal V_{set}. The total heat exchanger area A is 521.8 m^2 and the overall heat transfer coefficient U is assumed to be constant 26 W/(m^2K). The fluid in the circuit is a 34% glycol-water mixture (brine) and has a constant nominal mass flow $\dot{m}_{CT}$ of 2807 kg/h. The ambient temperature sensor T_{Amb} is installed near the CT for avoiding locational deviations in measurements. The temperatures entering (leaving) the CT from the CCM condenser circuit, T_{in} (T_{out}) are

Figure 1. (above) Microscale trigeneration plant at INES with Compression Chilling Machine (CCM) (below) Cooling tower (CT).

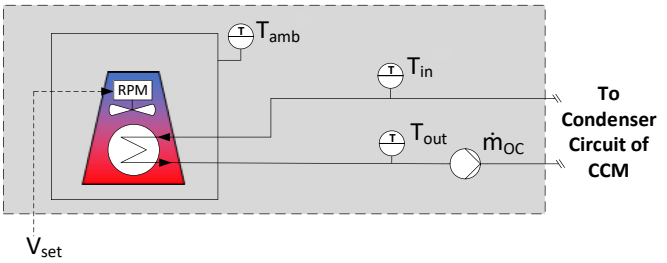

Figure 2. Controlled system.

measured with PT-500 sensors and have a dead time of approximately 1 minute.

A controller must achieve and maintain a setpoint T_{set} for the outlet temperature T_{out} and the control difference e is given by (1).

$$e = T_{set} - T_{out} \tag{1}$$

2.2. Cooling tower (CT) model

The CT model is motivated from the "Number of Transfer Units – Effectiveness (NTU-ε)" method (Bergman et al., 2011). Additionally, we applied the "Fan Affinity Laws" to establish the relationship between the RPM, the mass flow of air $\dot{m}_{air}$ and electrical power consumed by the CT P_{el} as seen in (3) and (4) (Wagner and Gilman, 2011). Here we assumed a directly proportional behaviour of fan speed with respect to the volt signal as seen in (2).

$$RPM = \frac{RPM_{max} V_{set}}{V_{set,max}} \tag{2}$$

$$\dot{m}_{air} = \frac{RPM \dot{m}_{air,max}}{RPM_{max}} \tag{3}$$

$$P_{el} = \frac{RPM^3 P_{el,max}}{RPM^3_{max}} \tag{4}$$

Other assumptions for this model are:

- Homogeneous air flow,
- Effect of the instantaneous variations of air speed on the pressure is neglected,
- No pressure loss over the heat exchangers.

The NTU-ε method calculates the effectiveness of a heat exchanger based on the maximum possible heat transfer that can be hypothetically achieved. The heat capacity rates for the hot (brine) and cold (air) fluids are denoted as C_h and C_c respectively and were calculated as shown in Equations (5(5) and (6)(6).

$$C_h = \dot{m}_{CT} c_{p,b} \tag{5}$$

$$C_c = \dot{m}_{air} c_{p,air} \tag{6}$$

Where, $c_{p,b}$ and $c_{p,air}$ are the specific heat capacities of air and brine respectively. And ṁOC and ṁair are their mass flows.

C_{min} (C_{max}) is the smaller (larger) out of the two heat capacity rates. The maximum possible heat transfer, $P_{th,max}$ for the CT per unit time was then given by (7).

$$P_{th,max} = C_{min}(T_{in} - T_{Amb}) \tag{7}$$

Applying (8), (9), and (10) the effectiveness of a cross-flow heat exchanger ε was calculated as follows:

$$NTU = \frac{UA}{C_{min}} \tag{8}$$

$$C_r = \frac{C_{min}}{C_{max}} \tag{9}$$

$$\varepsilon = \frac{1 - e^{[-NTU(1-C_r)]}}{1 - C_r e^{[-NTU(1-C_r)]}} \tag{10}$$

As shown in (11), ε was applied to calculate the thermal power P_{th} of the CT assuming it to be an air-fluid heat exchanger

$$P_{th} = \varepsilon \; P_{th,max} \tag{11}$$

By means of the above equations and with energy balance over the CT, the outlet temperature was calculated from (12).

$$T^*_{out} = T_{in} - \frac{P_{th}}{C_h} \tag{12}$$

The Information Flow Diagram (IFD) for the CT model is shown in Figure 3.

2.3. PID controller

A PID controller using the Ziegler-Nicholson method was developed. The following controlled system characteristics (averaged values) were used after performing a step-response analysis by changing the manipulated variable V_{set} from 0 to 10 Volts at different ambient temperatures ranging from ca. 24–28 °C.

- Time Constant T_S = 2.5 (min)
- Transfer Coefficient K_S = -2.2 (K/V)
- Dead Time T_t = 1 min

The characteristics of the controller were calculated as follows:

- Proportional Gain K_c = -1.63 (V/K)
- Integral-action Time T_i = 2 (min)
- Derivative-action Time T_d = 0.42 (min)

The control loop is seen in Figure 4. The controller was set-up in 40 person-hours with intermediate LabVIEW skills. Most time was needed for the step-response tests and no further tuning was done. A tolerance limit for e was implemented within which the V_{set} did not change to avoid excessive control actions.

2.4. Model based controller (MBC)

The MBC in this work is an open-loop controller combining the CT model and a rule-based algorithm. The flow-chart is seen in Figure 5.

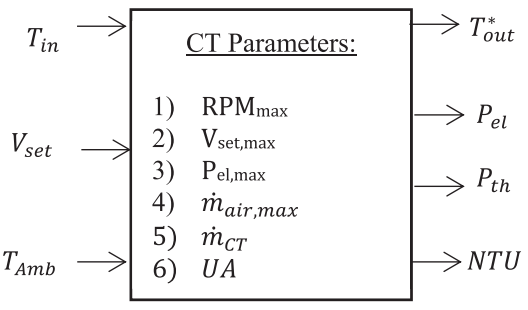

Figure 3. Information flow diagram of CT model.

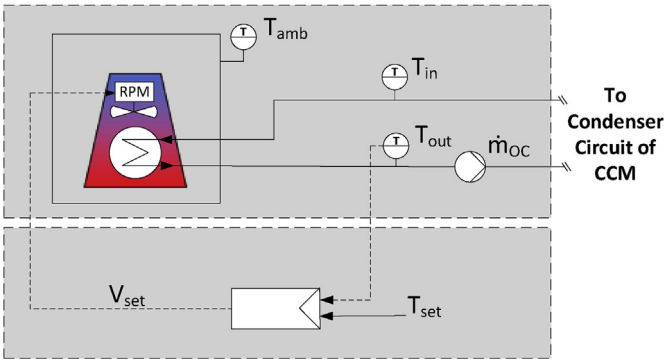

Figure 4. PID control loop.
Here, T_{set} - Reference variable (setpoint in fixed setpoint control)
T_{out} - Controlled variable (closed-loop control)
T_{Amb} and T_{in} - Interference variables
V_{set} - Manipulated variable.

Step 1. When magnitude of real error e is greater than tolerance limit the MBC is applied else the current V_{set} is applied further for duration of Control Horizon (CH) and then the loop is restarted.

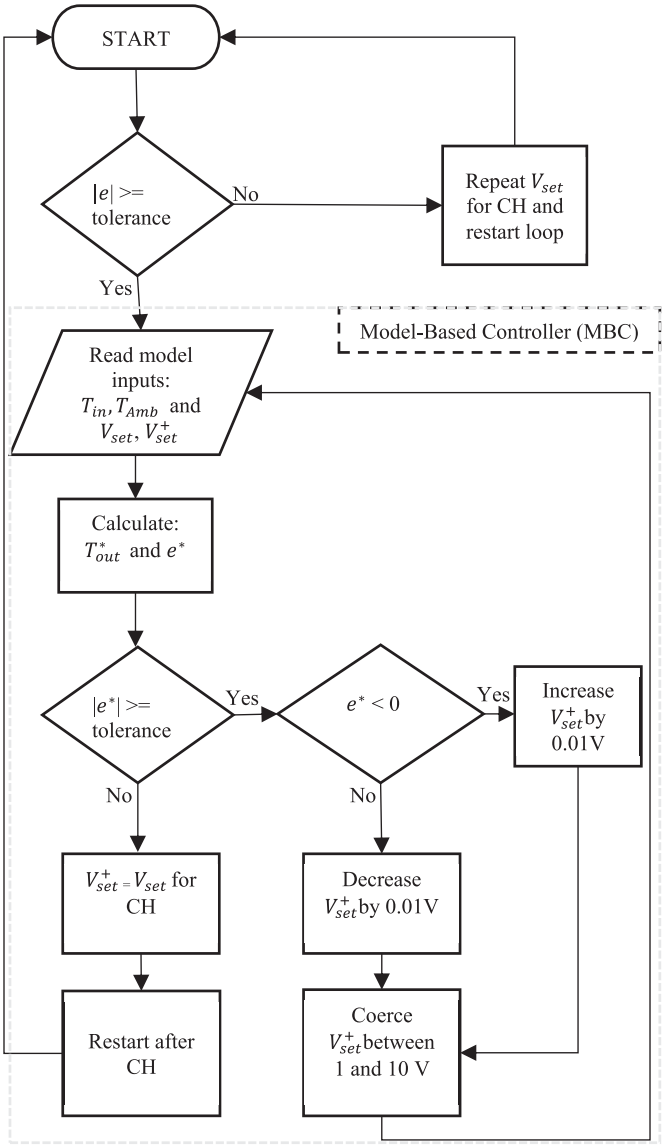

Figure 5. Flow-chart of the MBC.

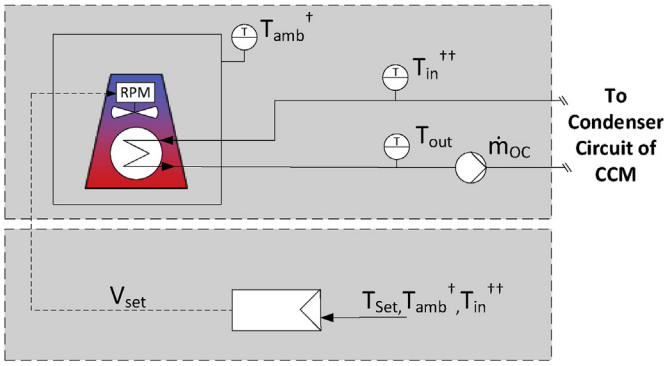

Figure 6. MBC control loop.

Step 2. The MBC uses current measurements of real T_{in}, T_{Amb} and V_{set} and the iteratively generated V_{set}^{+} as inputs to calculate T_{out}^{*} and corresponding simulated error e^{*}.

Step 3. When calculated e^{*} is greater than tolerance limit then Step 4 is followed, else the latest value of V_{set}^{+} used in Step 3 is applied to the plant as V_{set} for CH and the loop is restarted.

Step 4. If the e^{*} is negative then V_{set}^{+} used in Step 3 is increased by 0.01 V else it is decreased by 0.01 V. The new V_{set}^{+} is coerced between 1 and 10 V and applied in Step 2 again. Step 4 is repeated at 100 milliseconds to save processor power.

In this open-loop controller, the current value of the controlled variable is not fed back into the controller as shown in Figure 6. The controller was set-up in 0.45 person-hours with intermediate LabVIEW skills and no further tuning was done. The parameters necessary for the model were easily available in the data sheet of the CT.

3. Experiments and results

The controllers were tested under two different methods to compare their performance and efficiency. Firstly, under normal operation using fixed set-point control and secondly with step-response tests at varying ambient temperatures.

In the first method a set-point T_{set} of 30 °C was employed for the T_{out}. This value was used in accordance to the datasheet of the CCM (Daikin Europe, 2016). The reference controller was set-up to implement 10 V when the CT was on or 0 V when it was off which is often a standard practice in building HVAC, because a maximum RPM operation of the CT is expected in summer months. A tolerance limit of +/- 0.3 K was implemented for the PID and MBC. The CH for the MBC was set at 3 seconds.

In Figure 7, the operational comparison of the reference controller, PID, and MBC controls is shown. For a reasonable comparison, a 5 hour data set for each controller with similar operating conditions was chosen. The ambient temperature varied between 21 and 32 °C and cold tank temperatures were maintained between 25 and 26 °C. The reference controller cooled the T_{out} to the ambient temperature regardless of the T_{set} whereas, the PID and MBC reasonably maintained the set point. The T_{out} controlled by PID oscillated at lower ambient temperature and was more stable after T_{Amb} was higher than 26 °C while the MBC output stayed relatively smooth and continuous throughout. The electrical consumption of the CCM and CT together was 20.03 kWh_{el} for the reference controller, 17.20 kWh_{el} for PID, and 17.29 kWh_{el} for MBC over the 5 hours.

After consolidating data from multiple tests, the relationship between the manipulated variable V_{set} and interference variable T_{Amb} for the three controllers was established and is shown in Figure 8. Here, the average V_{set} of each controller is plotted against each T_{Amb}. The average is calculated using data points at +/- 0.5 K of the T_{Amb} and the standard deviation of the V_{set} represents spread of the control actions shown in the

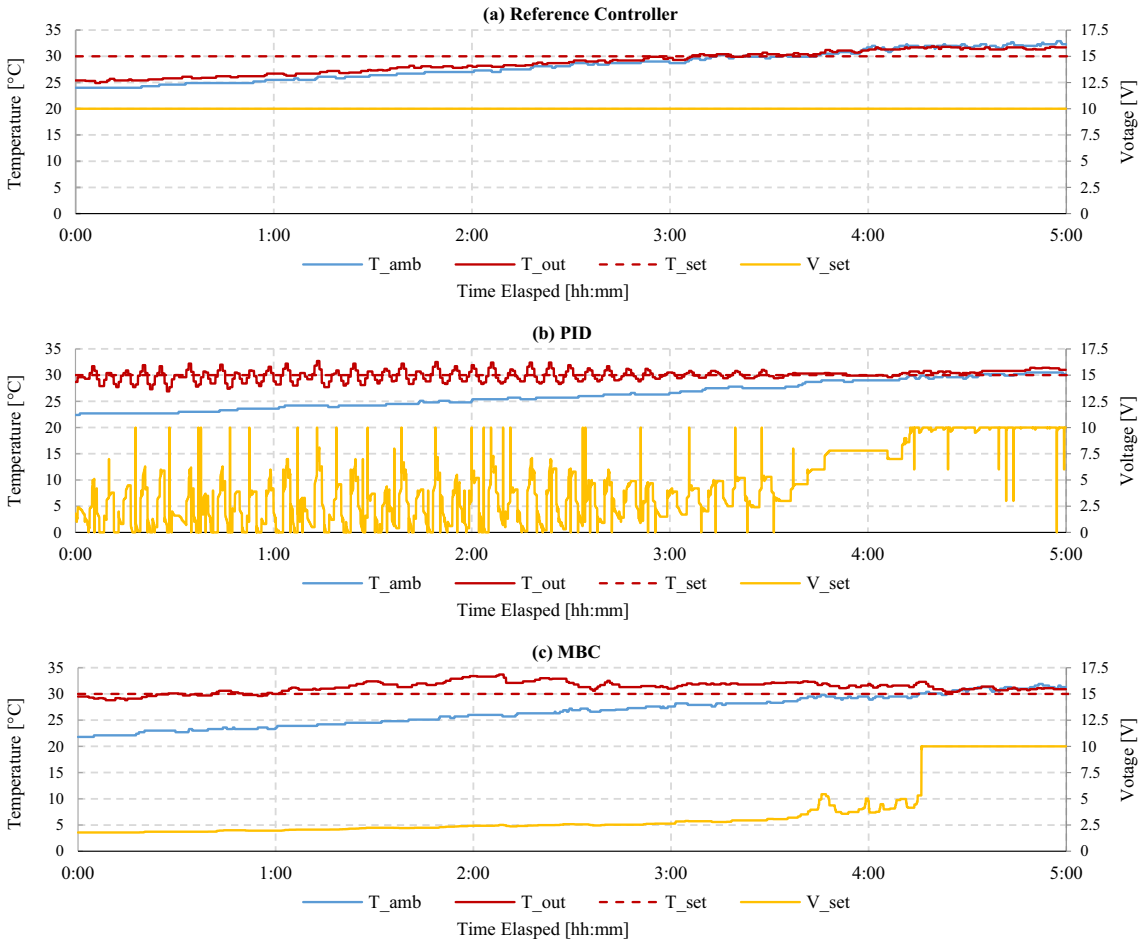

Figure 7. Sample data set for comparison of the three controllers. (a) The reference controller, (b) PID controller, and (c) MBC controller.

error bars. The PID controller shows steady increase in volt signal with T_{Amb} but it has a larger spread of control signal especially at lower ambient temperatures. This is in accordance with behaviour of PID V_{set} seen in Figure 7. The MBC volt signal for the CT also increases steadily with the ambient temperature but its magnitude tends to be lesser than the equivalent PID signal. Comparatively, the MBC has a smaller spread of control actions except at 30 °C since this includes data from 29.5 °C to 30.5 °C. At T_{Amb} below 30 °C the MBC significantly tries to control the CT to achieve the T_{set}.

However, at T_{Amb} above 30 °C the MBC calculates a 10 V signal and is steady thereafter. The PID takes control actions also at higher ambient temperatures seen in the spread of the data.

Additionally, the relationship between the average control difference and the T_{Amb} is plotted in Figure 9. The reference controller has the highest error at lower ambient temperatures and it decreases as T_{Amb} approaches 30 °C. The PID controller has a bigger spread but lower average error compared to MBC. The PID is most accurate when T_{Amb} lies between 27 and 29 °C. The error for MBC is positive at cooler T_{Amb} when slight overcooling occurs and becomes negative as T_{Amb} rises when undercooling occurs.

As T_{Amb} is greater than 30 °C a negative error is noticed for all controllers, since the CT cannot physically cool the working fluid below the T_{Amb}.

The second method examined the average time constant of the controllers to analyse their speed by doing step-responses. Case 1 was to introduce a disturbance by turning off the CT until T_{out} reached 40 °C and then turning the CT on with T_{set} at 35 °C. Case 2 was to run the CT on 10 V until T_{out} reached 25 °C and then turning the respective controller on

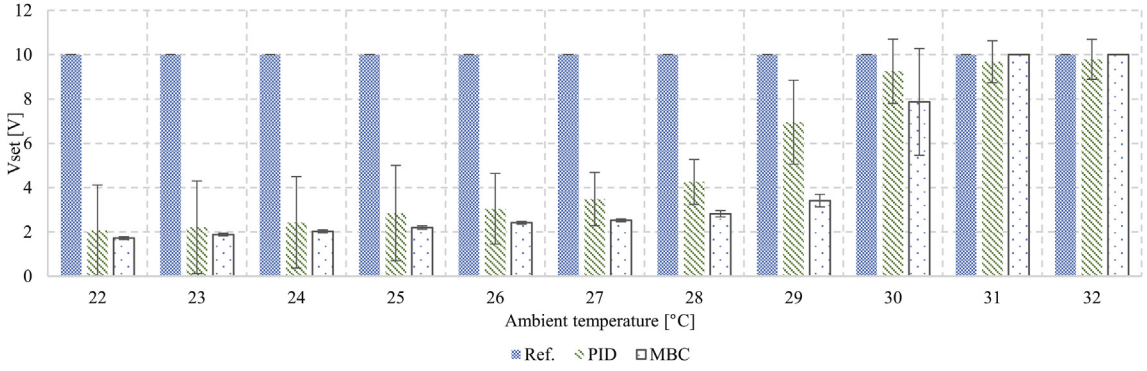

Figure 8. Relationship between the control signal and ambient temperature.

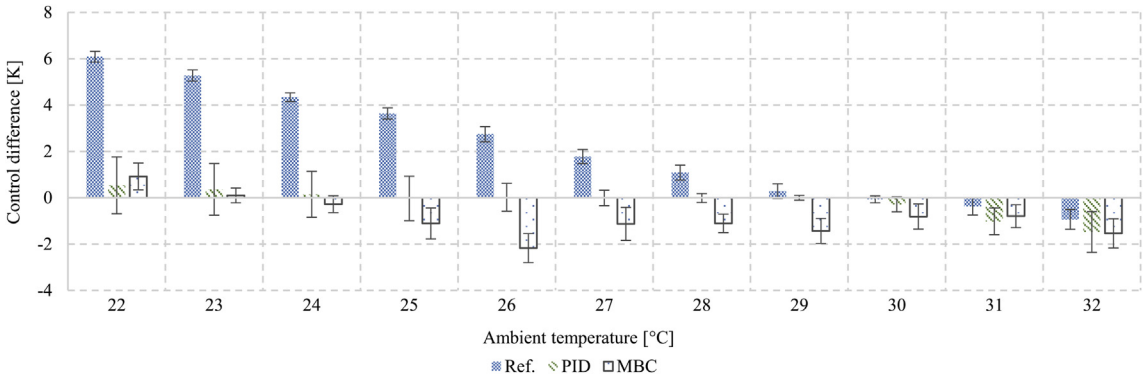

Figure 9. Relationship between the control difference and ambient temperature.

with T_{set} at 30 °C Table 2 shows the average results collected for multiple tests of the two cases. In both cases the PID was able to respond faster than the MBC by ca. one minute.

4. Discussion and conclusions

Three different controllers for a dry-cooling tower of a chilling machine were applied and compared in a realistic environment. A conventional controller was compared against more efficient PID and MBC controllers. The reference conventional controller used in this work was easy to implement and is often the standard practice in building technologies. In summer months with higher ambient temperatures it is appropriate for the CT to run at maximum speed to achieve a set-point close to the ambient temperature. However, in colder transition seasons energy is wasted as the medium is unnecessarily over cooled as seen in Figure 7. On the other hand, the PID and MBC controllers were more accurate and saved 14.5% and 14.1% electricity respectively by controlling the RPM or electricity consumption of the CT. A further saving in energy can be expected in transition seasons when ambient temperatures are cooler than the set-point of the CCM condenser inlet.

The PID was more complicated to set up but had higher accuracy. Its best accuracy was for ambient temperatures between 24 to 29 °C since it was set-up in this region. Outside this zone the controller deteriorated but was within acceptable limits of 30 °C +/- 2 K as recommended by the manufacturer of the CCM. This is in accordance with findings in literature regarding disadvantages of a PID controller operating outside its tuning region.

The MBC was very comparable to the PID as its control difference was also in the acceptable region and it had less fluctuations in controller output benefitting the CT hardware. Although the MBC was slower than the PID as seen in its larger time constants in Table 2 the thermal inertia of HVAC systems alleviates this delay. Additionally, a reduction in the control horizon for the MBC could improve its response time. The V_{set} for the MBC stays constant even when error increases as deduced by the smaller spread of data. This occurs since it is an open-loop controller and the assumptions in the CT model lead to inaccuracies. Its accuracy could be improved further by better parameterisation of the model, dynamic modelling of the system or extending it into a closed-loop controller. Both these steps are less complicated than tuning the PID for different operational ranges and for different CTs. Additionally, the parameters of the CT model are often available in manufacturer data sheets and it has been shown that the control logic can be implemented in the existing data acquisition and control system. This makes the MBC design more practical and it could be a potential improvement over PID since it is easier to set-up and generalise.

Experts can include above mentioned controllers over conventional On-Off methods either in green-field or retrofit scenarios to save operating costs and hardware degradation.

Table 2. Average time constant for different step-responses of controlled loop.

Case Type	PID	MBC
Case 1	131 s	219 s
Case 2	187 s	244 s

Declarations

Author contribution statement

Parantapa Sawant, Eric Ho & Jens Pfafferott: Conceived and designed the experiments; Performed the experiments; Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data; Wrote the paper.

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Competing interest statement

The authors declare no conflict of interest.

Additional information

Data associated with this study has been deposited at Mendeley under the accession number https://doi.org/10.17632/ytwjtc2sr7.1.

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